MECHANICAL

ENGINEERING MATERIALS:

THEIR PROPERTIES AND TREATMENT IN CONSTRUCTION.

BY

EDWARD C. R. MARKS,

ASSOC. M.I.C.E., M.I. MECH. E.,

Lecturer on Engineering at the Birmingham Municipal Technical School.

PRICE 1s. 6d. NET.

1893.
THE TECHNICAL PUBLISHING COMPANY LIMITED,
6, VICTORIA STATION APPROACH, MANCHESTER.
JOHN HEYWOOD,
DEANSGATE AND RIDGEFIELD, MANCHESTER.
2, AMEN CORNER, LONDON, E.C.
33, BRIDGE STREET, BRISTOL.
And all Booksellers.

PREFACE.

In presenting the series of articles on "Mechanical Engineering Materials," originally appearing in *The Practical Engineer*, in their present form, the Author is hopeful that they may be found useful to engineers and to students with some experience in practical engineering. They embrace the requirements of the City and Guilds of London Institute in a great part of one section of their Annual Examination in "Mechanical Engineering."

To very young students, and those unacquainted with actual engineering work, the pages will probably be in a great measure unintelligible; but the Author has endeavoured to impart reliable and concise information to readers who, in their daily vocations, have acquaintance with the subjects discussed in this little volume.

E. C. R. M.

13, Temple Street, Birmingham,
August, 1893.

CONTENTS.

INDEX.

MECHANICAL ENGINEERING MATERIALS.

INTRODUCTION.

THE strength and general properties of the materials of construction employed in all branches of engineering enterprise have received very careful investigation from time to time by engineers of great ability and high rank in the profession. The more important results of their labours have been published in valuable treatises on the subject, or have been made public by papers read before various learned societies, and printed in the Transactions of such bodies. At the present day investigation and research are still proceeding, both in the direction of the introduction of new materials, and also in the endeavour to improve old and known materials by the elimination of impurities, the introduction of other substances into the body, or by the establishment of better methods of working, following from a more thorough understanding of the physical properties of the material under various conditions.

In the meantime, however, with all the contemplated changes and improvements in materials, and in the methods of adapting same for public service, it is, of course, necessary for the practical engineer to proceed with his labours, employing the best material and the best known methods in accomplishing his purpose. It is our purpose, therefore, in a short series of articles, to give concise notes on the strength and properties of existing materials that come under the treatment of the mechanical engineer in machine and general constructive work. The particulars given will be essentially practical in character, and whilst primarily intended for the younger or student class of our readers, it is believed that the notes will be found of general interest.

Iron and steel, as the most largely employed materials, naturally call for first consideration. Of iron we have the two chief classes, cast iron and wrought iron, whilst of steel we have many classes. Before proceeding to treat of cast iron in detail, we may note that although there is such a

great difference in the physical character of the different varieties of steel, wrought iron, and cast iron, yet in their chemical composition there is not so great a difference as might be expected. They each consist almost entirely of the pure metallic element—iron, with the addition of small quantities of other elements (but chiefly the element carbon). Chemical analyses will be given later, and it will be seen as we proceed that a minute fraction of 1 per cent of another element will, in many cases, make a great difference in the physical properties of the metal.

CHAPTER I.

Cast Iron.

The valuable feature of this metal to the engineer consists chiefly in the ease with which he can form it into complicated shapes, either of large or small bulk, by moulding from the pattern and running the molten metal into the mould so formed. The iron reaches the founder in the form of "pigs," produced at the blast furnaces in the various iron-melting districts. Such pigs are classified for commercial purposes, both by numbers and also by the districts in which they have been produced. But all pig iron may be classed as either grey, white, or mottled. The grey is a soft quality, deficient in strength, but very fluid when melted; the white is hard and strong; whilst the mottled, as its name somewhat implies, is a medium between the white and grey. Grey iron and mottled are used for foundry purposes; the hardest, white iron, is employed only for conversion into wrought iron and steel.

There are many varieties of grey and white iron, and a skilful founder will mix the varieties before melting in the cupola, to suit the nature of the castings he has to make. In too many foundries, however, the practice is to buy in a quantity of pig iron at the cheapest possible rates, melt it down in the cupola, and use it indiscriminately for all classes of work. Castings that have to be machined up, but in which strength is not an important point, may be made of a soft grey quality, easily tooled, whereas castings that have to possess considerable strength must be produced from harder varieties, or mixtures of different kinds of pig.

Cast iron consists essentially of pure iron and carbon, the amount of carbon varying from about 3 to 4 per cent. The chemical difference between white and grey iron is that in

the former case the carbon present in the metal is all chemically combined with the iron; but in grey iron only a portion of the carbon is chemically combined, the other portion being merely mixed with the iron. The more free

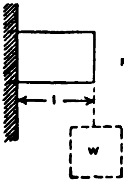


FIG. 2

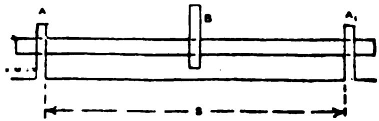


FIG 1

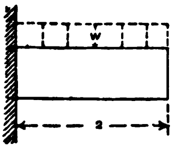


FIG 3

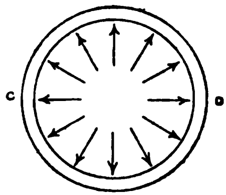


FIG 6

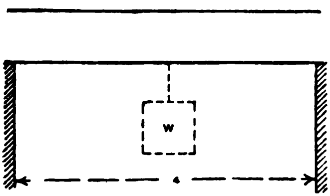


FIG. 4

FIG. 7

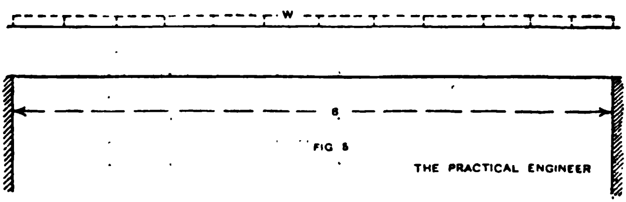


FIG 5

carbon, the greyer will be the iron. Manganese, silicon, and other elements also occur in varying small percentages in all varieties of cast iron.

The strength of cast iron, as demonstrated by the mechanical testing machine, will vary, of course, with the nature of the metal, but, as a general average result, the ultimate or breaking tensile strength of good commercial cast iron may be taken at 7 tons per square inch, and the ultimate crushing or compressive strength at 40 tons per square inch. The working load will depend as to whether the material will have to resist a steady or varying load; in the former case the working load may be $\frac{1}{6}$th of the breaking load, but when subjected to shocks or impact a factor of safety of 10 should be employed.

Owing to the low tensile strength of cast iron, it is not usual to employ it as a tie, but to confine it as far as possible to the resistance of compressive stresses. With due care in the selection and assortment of pigs, cast iron can be produced that will give an ultimate tensile strength of 15 tons per square inch, but this is a result not to be looked for with the ordinary castings of daily manufacture.

Test Bars.—With all castings in which strength is an important feature, test bars should be run from the same metal as that employed in the castings, and these bars should be afterwards tested by loading them as a beam. The size of the bar is generally 3 ft. 6 in. long, and either 2 in. deep by 1 in. in breadth, or of the same length, and 1 in. square section. The bar is tested as a beam, supported firmly at each end, with a load applied at the centre. Fig. 1 of the adjoining sketches represents a test bar sustained by the supports A A^1, whilst the load is applied from a dead weight or other testing machine by means of the bridle B. The following table gives results of tests upon test bars of cast iron employed in the manufacture of socket and spigot water pipes :—

Number of Test.	Section of Bar.	Load carried in Centre (supports 3ft. apart).	Deflection of Bar before Fracture.
1	1 in. square	700 lb.	·32 in.
2	1 in. „	820 lb.	·322 in.
3	1 in. „	1000 lb.	—
4	1 in. „	970 lb.	—
5	1 in. „	950 lb.	—
6	1 in. „	900 lb.	—
7	2 in. × 1 in.	3400 lb.	—
8	2 in. × 1 in.	3500 lb.	—

In the case of Nos. 7 and 8, the deep (2 in.) side was set vertically, so that the bar should by the principles governing the strength of beams (which we next proceed to

investigate) have resisted a load four times as great as that carried by the 1 in. square bar. It will be noted that the breaking strength given in table for Nos. 7 and 8 amounts to very nearly four times the average breaking strength of the 1 in. square bars.

A very useful formula for calculating the breaking strength of solid cast-iron beams, loaded at the centre, and supported at each end, is as follows :—

$$W = \frac{B\,D^2}{L}$$

where W is the breaking load in tons, B the breadth of the beam in inches, D the depth of the beam in inches, and L the length or distance between the supports in feet. Let us proceed to verify or test the accuracy of this formula by the information obtained from the experiments. Take the case of a 1 in. square beam or bar 3 ft. between the supports, then

$$W = \frac{1 \times 1}{3} = \tfrac{1}{3} \text{ of a ton} = 746\tfrac{2}{3} \text{ lb.}$$

It will be seen, on reference to the table, that in one case only did the bar fail to carry the breaking load as given by formula. In the same manner we can apply the formula to the case of a bar 2 in × 1 in., as follows :—

$$W = \frac{1 \times 4}{3} = 1\tfrac{1}{3} \text{ ton} = 2986\tfrac{2}{3} \text{ lb.,}$$

and it will be observed that both test pieces recorded in table give a considerably higher breaking load.

If, instead of the load being applied at the centre of the bar, it is evenly distributed throughout its length, then the breaking load will be twice the amount of the centrally applied load ; or if the same load that causes breakage of a beam when applied at the centre of its length is evenly distributed, then the length of the beam may be doubled. In the same manner, when the load is applied to the end of a beam or bar supported at one end only, the length of the beam must be only one-quarter the length of the central loaded beam supported at each end ; whilst if the load is evenly distributed on a beam supported at one end only, the length may be one-half that of the central loaded beam supported at each end.

Figs. 2, 3, 4, and 5 show the methods of loading the beams and their relative lengths when the load to be carried W

(represented by dotted lines in each case) is equal, and the breadth and depth of the beams also being the same.

The formula for each case is as follows :—

Fig. 2 $$W = \frac{B\,D^2}{4\,L}.$$

Fig. 3 $$W = \frac{B\,D^2}{2\,L}.$$

Fig. 4 $$W = \frac{B\,D^2}{L}.$$

Fig. 5 $$W = 2\frac{B\,D^2}{L}.$$

The letters represent the quantities previously given.

It must be clearly understood that these formulæ give the *breaking* or *ultimate loads.* Under steady loads the working strength may be taken at $\frac{1}{6}$th of the breaking load, but in resisting a variable and suddenly applied load a higher factor of safety must be employed.

The results given in table of breaking loads on test bars no doubt exceed those that would be obtained with tests upon much of the cast iron now employed, but in all works of importance it should be specified that test bars run at the same time as the castings should give results in no case less than given by the formulæ verified by our table.

The strength of a pipe or a cylindrical vessel to resist an internal bursting pressure will vary inversely as its diameter and directly as the thickness of its metal; that is to say, a pipe having a diameter of 20 in. will require to have twice the thickness of metal employed in a 10 in. pipe, in order that they may be of equal strength. But this rule cannot be strictly adhered to in practice, because the smaller pipes have to be made with a greater thickness of metal than actually required for strength, or otherwise they could not be successfully moulded or cast soundly. As an example, let us assume that a 20 in. internal diameter cast-iron pipe has to be made and is to be employed as a water-pipe line where the pressure is 100 lb. per square inch. Before passing into service, the pipe should be tested by hydraulic pressure to twice the working pressure, which in this case will be 200 lb. per square inch. Now, the effect on the interior of the pipe is represented by the radial arrows in fig. 6. On every square inch of the internal surface there will be a pressure of 200 lb. when the pipe is under test pressure.

But if all these radial forces were resolved into direct horizontal and vertical forces by the principle of the triangle of forces, as illustrated in fig 7, it would be found that the total tearing or tensile force acting on either side of any diametrical line, say a line drawn through C D, would be represented by P × D, P being the pressure in pounds per square inch, and D the diameter of the pipe in inches.

Thus, with a 20 in. pipe and 200 lb. pressure, the force to be resisted by the metal at A and B will be 20 × 200 = 4,000lb. If the cast iron has a tensile strength of not less than 7 tons, we may take its safe strength under test load to be 1½ tons per square inch, and therefore, to resist 4,000 lb., we shall require 4,000 ÷ 3,360 = 1·19 square inches. But as it is impossible in ordinary casting to ensure that the metal shall be of exactly equal thickness on either side of the diameter, the thickness of the metal should be not less than $\frac{11}{16}$ in.

CHAPTER II.

Cast Iron *(continued).*

Hydraulic Pipes.—In calculating the strength of cast-iron pipes for conveying water or other liquid under the moderate pressure existing in town mains or similar pipe lines, we may safely employ the simple formula given in the previous chapter. Each formula, it will be noted, assumes that double the strength is obtained for double the thickness of metal employed. This assumption, however, is not strictly accurate, for when a pipe or other cylindrical vessel is subjected to internal pressure, the iron undergoes a certain amount of extension or stretching, some slight extension taking place with every increment of pressure. But the interior surface or circumference of the pipe or cylinder will suffer a greater amount of extension than the exterior surface or circumference, thus demonstrating that the greater stress is brought to bear upon the inside surface of the metal, and the stress decreases in intensity as we approach the exterior surface.

In calculating the strength to be given to the cylinders of hydraulic presses which have to resist pressures up to 2 tons and 3 tons per square inch, it is important that this unequal distribution of the bursting stress should receive due consideration. Barlow, in his valuable treatise "On the Strength of Materials and on Construction," published as early as 1837, discusses this matter at considerable length, and gives

the following rule for computing the thickness of metal to be given to the cast-iron cylinders of hydraulic presses: "Multiply the pressure per square inch by the radius of the cylinder, and divide the product by the difference between the cohesive (or tensile) strength of a square inch rod of the metal and the pressure per square inch; the quotient will be the thickness required." Thus, for example, suppose that we wish to cast a cylinder, 20 in. diameter, for a hydraulic press which is to withstand a working pressure of 2 tons per square inch. The cast iron employed should be of such quality that test bars produced therefrom will be capable of withstanding a tensile stress of 9 tons per square inch before fracture. But the greatest steady load that may be put upon a 1 in. section, without injury to the metal, must not be considered as more than $3\frac{3}{4}$ tons; therefore the thickness required will be

$$\frac{10 \times 2}{(3\frac{3}{4} - 2)} = \frac{20}{1\frac{3}{4}} = 11\frac{1}{2} \text{ in.}$$

This cylinder could be tested to 3 tons per square inch without injury. The results obtained by the application of Barlow's rule generally agree with modern practice in hydraulic presses for pressures of about 2 tons per square inch.

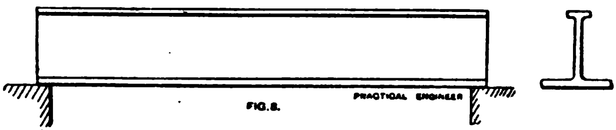


FIG. 8.

The following simple formula is sometimes employed for ascertaining the thickness of cast-iron cylinders for hydraulic work to withstand any given *test* pressure:—

$$T = \frac{\text{radius of cylinder} \times P}{3}$$

T = thickness of cylinder walls in inches.

P = test pressure in tons per square inch.

Radius of cylinder to be taken in inches.

This rule may be applied with safety for test pressures of 1 ton to $1\frac{1}{2}$ tons per square inch, but not higher pressures.

Working pressure must not exceed two-thirds of test pressure for hydraulic presses, and in some cases, such as hydraulic lift cylinders, the working pressure does not exceed one-fourth of the test pressure.

Cast-iron Flange and Web Girders.

In practical machine design and construction but comparatively few instances occur where a beam or girder of cast iron is employed of a solid rectangular section. The rules given in our last chapter will, however, be constantly found useful, not only in calculations as to the strength of test bars, but also for application in calculations where cast iron is employed in the cantilever girder form. Thus the arms of a flywheel or pulley are examples of cast-iron cantilevers; the boss or hub forms the support, whilst the weight or load is applied at the rim end of the arms.

When a cast-iron girder of considerable dimensions is to be constructed, the rectangular or other solid section is discarded and replaced by a flange and web girder, as shown in fig. 8. With girders supported at each end, such as the solid rectangular section girder shown in fig. 9, the under side has to resist a tensile stress, whilst the upper side is in compression. Starting from the bottom side A of the

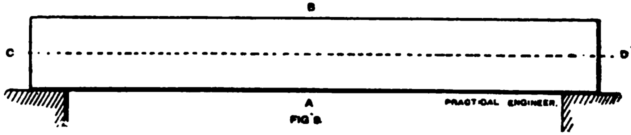


FIG. 9.

girder and moving towards the upper side B, we should find that the tensile stress on a loaded girder would become less and less, until at length we should pass into the region of the compressive stress, and such compressive stress would increase in intensity as we proceeded towards the upper side B. At some point between A and B it is evident that we must have crossed a boundary line or line of demarcation between the compressive stresses on the one side and the tensile stresses on the other. This line is, of course, quite imaginary, but it is a fact that in every girder there is a position somewhere between the upper and lower sides where the metal or material is neither in tension nor compression. Hence a line drawn along the girder indicating the probable position of this boundary line of the two kinds of stresses is termed the *neutral axis* of the girder.

If the beam or girder is formed of metal having equal strength in tension and in compression, the neutral axis will pass along the girder at about its mid-depth, as shown by dotted line C D in fig. 9. But if the metal be weaker in tension than in compression, as cast iron, for instance,

then the neutral axis will approach nearer to the compression side of the beam.

From the foregoing description it will, we think, be clearly understood that the metal in the neighbourhood of the neutral axis offers very little resistance to the stresses set up by the load, but that the great intensity of stress is at the top and bottom surfaces of the beam. We may, therefore, cut away a considerable portion of the metal near the neutral axis without materially weakening the beam. Thus the beam or girder represented by the sectional end view, at fig. 11, is almost as strong as a beam of same depth and width, but of solid section, as shown at fig. 10. But if now, with the same amount of metal employed in casting a solid beam, such as fig. 11, we construct a girder having such a form as shown by the sectional view in fig. 8, having

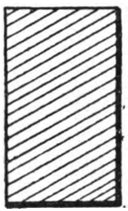
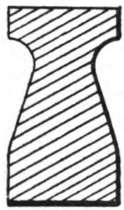


FIG. 10.



Practical Engineer. FIG. 11.

two flanges and a connecting web, then, although of the same weight, yet the flange and web girder will sustain a much greater load than the solid beam.

Since the introduction of rolled iron and steel joists or girders, cast-iron girders are not very largely used in constructive purposes. It should be remembered, however, that the principle of the division of the stresses by the neutral axis, as herein discussed, applies to girders constructed of *any* material. We shall take examples showing the ascertainment of the strength of rolled iron and steel joists in due course.

Since ordinary cast iron is about six times stronger in compression than in tension, a flange and web girder, designed on theoretical principles, should have its compression flange only one-sixth the area of the tension flange. Thus, suppose we require a cast-iron girder to support a central load of 2 tons; span of girders, or distance between supports, to be 12 ft.; mean depth of girder

(measured between the centres of top and bottom flanges) to be 1 ft.; then the stress on the top and bottom flanges at the centre of the girder will be—

$$\frac{\frac{1}{2}\text{ load} \times \text{half span}}{\text{depth}} = \frac{1 \times 6}{1} = 6 \text{ tons.}$$

Taking the safe tensile strength as $1\frac{1}{2}$ tons per square inch, then the required area of the tension flange (the bottom flange in this instance) amounts to—

$$\frac{6}{1\frac{1}{2}} = 4 \text{ square inches.}$$

and $\frac{4}{6} = \frac{2}{3}$ square inch = area of compression flange.

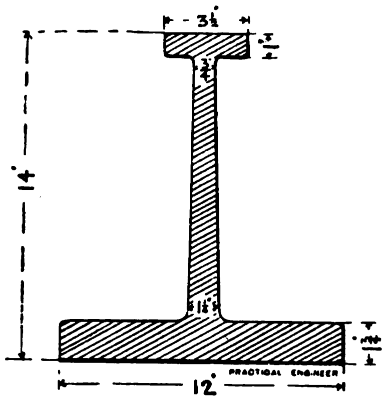


FIG 12

It is frequently necessary in practice to make the compression flange of greater area than the calculated amount, in order to obtain uniformity of section. A girder having considerable variation of section, especially in the thickness of the metal, is difficult to successfully mould and cast. Moreover, there is a buckling or crippling stress in the compression flange, which has to be provided for in girders of long span.

The most simple and reliable formula that can be employed to ascertain the ultimate or breaking central load of a cast-iron girder is the following:—

Breaking load in tons = 2 (area of bottom flange × 7 × mean depth of girder ÷ half span of girder).

In this formula the strength is calculated from the tension flange only. The compression flange must be not less than one-sixth the area of the tension flange.

Fig. 12 represents a section of a cast-iron girder in which the area of bottom flange is 21 square inches, the top flange being one-sixth area of bottom flange. In the "Strength of Materials" (by Box) the results of experiments on thirteen girders, of section given in fig. 12 are recorded. The girders were of 16 ft. span, loaded in their centre, supported at each end. They were cast from various kinds of common British cast iron. The minimum breaking load of girders was 30 tons, and the maximum $47\frac{1}{4}$ tons; mean breaking load of the thirteen girders = 38·3 tons.

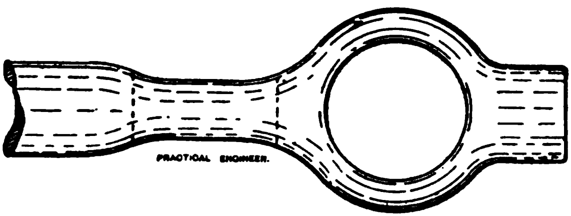


FIG. 13.

By applying our formula, we obtain the following result:

$$\text{Breaking load} = 2\left(21 \times 7 \times \frac{12\frac{5}{8}}{96}\right) = 38{\cdot}6 \text{ tons.}$$

It will be seen that this result is almost identical with the average result obtained from actual experiment.

CHAPTER III.

WROUGHT IRON.

THE elimination of the small quantity of carbon, and the still smaller quantities of the elements sulphur, phosphorus, silicon, and manganese, from cast iron, presents us with the material known as wrought or malleable iron. Malleable iron is the text-book name for wrought iron, and the practical man is liable to some little confusion over the term, for the malleable iron, as known by the mechanical

engineer, is malleable cast iron, a material we shall presently consider. In these articles wrought iron will be known by no other name.

Wrought iron is usually sent out from the mills in the form of plates, or of bars of varying section. Specifications for such plates or bars generally contain a clause to the effect that the iron shall contain not more than ·05 per cent of carbon, or 0·01 per cent of phosphorus, sulphur, silicon, or manganese. But a chemical analysis does not give entirely satisfactory evidence as to the quality of wrought iron, and some engineers even consider such test to be useless. The amount of impurity in the iron may be so small as to be almost inappreciable to the chemist, but yet its presence will make a vast difference in the physical properties of the metal.

The ultimate or breaking tensile strength of good quality wrought-iron bars should not be less than 22 tons per square inch, with an elongation in an 8 in. length test piece of not less than 20 per cent, and contraction of area at fracture not less than 40 per cent. With wrought plates of first-class quality, such as boiler plates, the tensile strength will be as high as for bars, but the elongation and contraction will be less.

The following table gives results of tests on special quality wrought bars of Staffordshire manufacture :—

Description of bar. In. In.	Breaking load per square inch of original area. Tons.	Contraction of area at point of fracture. Per cent.	Elongation in 10 in. length. Per cent.
$2\frac{3}{4} \times \frac{5}{8}$	22·31	31·43	28·5
$2 \times \frac{3}{4}$	23·27	46·64	29·5
$3 \times \frac{7}{16}$	23·70	30·12	25·6
$1\frac{3}{8}$ round	22·30	49·57	32·75
$\frac{5}{8}$ ”	22·96	45·47	24·8
$\frac{1}{2}$ ”	25·52	48·4	22·7

Specifications for iron boiler plates usually call for 15 per cent contraction of area at fracture when tested along the grain or in direction of fibre, and $7\frac{1}{2}$ per cent when tested across the grain. With some plates there will be a marked difference between the tensile strength in direction of fibre and the strength across the fibre. In the table on page 14, however, which gives results of tests on a number of samples cut from iron boiler plates manufactured in Shropshire, it will be seen that in many instances the strength across the fibre is greater than the strength in the direction of the fibre.

Two very common defects in wrought iron are those known by the terms "hot-short," or "red-short," and "cold-short." Hot-short iron is difficult to forge, whilst cold-short iron will not withstand hammering or bending when cold.

TESTS OF ⅝ IN. IRON BOILER PLATES.

No. of plate.		Breaking load. Tons.	Contraction of area at point of fracture. Per cent.	Elongation in 8 in. length. Per cent.
1	W	22·50	15·3	11·0
	A	23·45	17·0	10·5
2	W	22·45	15·5	11·3
	A	22·50	13·8	11·5
3	W	23·25	18·6	8·8
	A	22·90	14·5	6·5
4	W	22·85	15·8	9·0
	A	23·05	14·5	11·2
5	W	22·00	12·2	8·0
	A	21·50	12·3	5·5
6	W	23·10	21·5	13·5
	A	23·00	14·0	8·5
7	W	22·60	19·4	14·0
	A	23·25	17·8	13·0
8	W	22·90	9·6	7·0
	A	22·00	12·9	7·0
9	W	22·45	18·5	12·0
	A	23·15	13·4	8·0
10	W	21·90	16·0	10·0
	A	22·30	15·7	10·0
11	W	22·80	21·6	11·5
	A	23·15	18·0	8·0
12	W	23·10	16·5	11·2
	A	22·95	16·3	10·5
13	W	22·95	15·8	10·0
	A	23·20	16·4	13·0
14	W	22·90	10·9	11·0
	A	23·35	14·6	10·0
15	W	23·70	19·0	12·5
	A	23·70	16·9	13·0

To ascertain if iron is free from these defects it is subjected to tests as follow :—

Hot Test.—The iron sample bar is heated to a good red heat, and a hole is then punched through it at a distance of about one and a half diameter from the end of the bar ; the hole is then drifted out to one or one and a half diameter

of the bar, according to the quality of the material. A second hole is then punched and drifted at right angles to the first. The appearance of the bar after this test will be as shown in fig. 13. The extreme end of the bar may now be "ramshorned" by splitting the metal through A A, and forging and bending back, as shown in fig. 14. The heat at which such tests can be best performed will depend on the brand of the iron; some qualities can stand much more fire than others. If the iron is quite free from hot-shortness, there will be no cracking or sign of fracture at the edges of the metal around the holes.

Cold Test.—The bars must be able to stand being bent double, in cold state, over a rod of same diameter as their

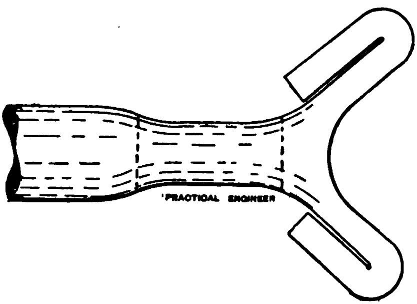


FIG. 14.

thickness, without fracture. If the bar is of much greater width than thickness, it will be found more rigid, and the cold test, as above, cannot be strictly adhered to. Another good cold test is to notch the sample with a chisel, and bend it back, as shown in fig. 15. This will reveal the fracture, which should have a fibrous appearance, and be free from cinder, &c.

The resistance of wrought iron to a compressive or crushing stress has not been subjected to so many tests as its resistance to tensile stress. In simple compression, however, its resistance may be taken as equal to its tensile strength. In the same manner the shearing and the torsional or twisting strength may generally be considered to be but little less than the tenacity.

Wrought-iron Shafting.

Probably the chief application of wrought iron to resist a torsional stress is to be found in the employment of shafting for the transmission of power. In short shafts we have simply to take into account the resistance of the material to a direct torsional stress, but in long shafts we must also consider the provision of necessary stiffness. The general principles governing the application of short-length shafts may be summed up in two very simple rules (the proof of which will be evident as we proceed) as follows: (1) The power that can be transmitted by any shaft increases in direct proportion to increase of its speed. (2) The strength of a shaft varies as the cube of its diameter. Carrying these two rules in our mind, we have only to ascertain the capability of a certain standard shaft running at a given speed, in order to readily obtain the service of which a shaft of any other diameter and speed is capable.

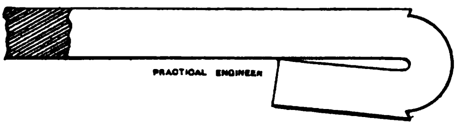


Fig. 15.

It has been found by practical experiment that a 1 in. diameter wrought-iron shaft has such ultimate torsional strength as to require a weight of about 700 lb. hanging at the end of a lever 1 ft. long in order to cause fracture. In fig. 16 such a lever and weight are represented by dotted lines.

The stress set up by the weighted lever on the shaft is in reality a torsional or shearing stress, for the lever may be considered as pivoted at the centre of the 1 in. diameter bar or shaft, and the resistance counteracting the effect of the weight is provided by the torsional strength of the material acting at a mean distance of $\frac{1}{4}$ in. (or one-half radius) from the centre of the bar. The amount of metal to be sheared through is the circumference of the circle of resistance multiplied by the radius of the bar, and this in the case before us will be

$$\tfrac{1}{2} \times \frac{22}{7} \times \tfrac{1}{2} = \frac{22}{28} = \frac{11}{14} \text{ square inch.}$$

Allowing the ultimate shearing strength to be 20 tons per square inch, then the ultimate strength of the 1 in. shaft will be

$$\frac{11}{14} \times 20 = 15\tfrac{5}{7} \text{ tons} = 35{,}200 \text{ lb.}$$

We have seen that the leverage of resistance is only $\frac{1}{4}$ in., but the leverage of the weight is 12 in., giving a ratio of 48 : 1, and 35,200 divided by 48 gives us 733 lb. as the weight required at the end of the 1 ft. lever according to calculation.

If, now, we take a 2 in. shaft, then the area to be sheared becomes—

$$1 \times \frac{22}{7} \times 1 = 3\tfrac{1}{7} \text{ square inches};$$

and, therefore, the shearing force required will be—

$$3\tfrac{1}{7} \times 20 = 62\tfrac{6}{7} \text{ tons} = 140{,}800 \text{ lb.}$$

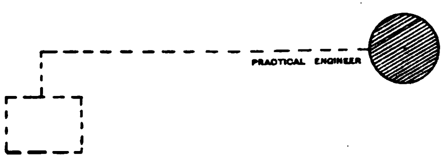


Fig. 16.

But the ratio of leverage of weight to leverage of resistance is now only 24 : 1; therefore the weight required at end of 1 ft. lever is 140,800, divided by 24, giving us 5,866 lb., or eight times (or 2 cubed) the amount required for the 1 in. shaft. Hence it will be seen that the strength of a shaft varies as the cube of its diameter.

It must be clearly understood that we have been calculating the ultimate or breaking strength of shafting. In practice a factor of safety of 6 should be employed.

The horse power that can be transmitted by a shaft varies with the speed of the shaft, because an increase of speed, within reasonable limits, does not add any torsional stress to the shaft, such stress depending simply on the weight to be turned or revolved by the shaft, and its distance from the centre of same.

Having considered the principle governing the great addition of strength following on an increase in diameter,

we may now express the ultimate torsional strength of a wrought-iron shaft by the following simple formula :—

$$W = \frac{700 \times D^3}{R}$$

W = ultimate load in pounds applied at rim of pulley secured to shaft.
D = diameter of shaft in inches.
R = radius of pulley in feet.

As an example we will take the case of a 3 ft. 6 in. belt pulley, secured to a $2\frac{3}{4}$ in. shaft; we are required to find the pull or tension at the rim of the pulley that can be safely resisted by the shaft. Applying our formula, we have—

$$W = \frac{700 \times (2\frac{3}{4})^3}{1\frac{3}{4}}$$

$$= \frac{700 \times 20{\cdot}8}{1{\cdot}75}$$

$$= 8320 \text{ lb.}$$

Allowing a factor of safety of 6, then the safe pull or tension at rim of pulley becomes—

$$8320 \div 6 = 1386 \text{ lb.}$$

If the shaft is running at 50 revolutions per minute, then the horse power that can be transmitted will be—

$$\frac{1386 \times \text{circumference of pulley in ft.} \times \text{revolutions per min.}}{33000}$$

$$\frac{1386 \times (\frac{7}{2} \times \frac{22}{7}) \times 50}{33000}$$

$$= 23{\cdot}1 \text{ H.P.}$$

It will be observed that if the speed be doubled, then the horse power that can be transmitted will be doubled also.

With long lines of shafting, although we may have sufficient torsional strength, yet trouble may be experienced by jerky and unsteady running if sufficient torsional stiffness is not provided. Hence we find that shafting of great length is of larger diameter than our rule for torsional strength would indicate.

Shafts over $4\frac{1}{2}$ in. diameter which have sufficient torsional strength for the work required of them will generally be found to have also sufficient stiffness, but under that

diameter an increased area must be given to long line shafts, such increase varying with the length of the line. It is important to note that whereas the torsional strength of a shaft increases as the third power, or as the cube of the diameter, the torsional stiffness increases as the fourth power of the diameter, or as the square of the area. Thus, a 4 in. diameter shaft will transmit sixteen times the force that can be transmitted by a 2 in. diameter shaft without undergoing any great degree of twisting. It must be remembered also that although in calculating the torsional strength no consideration is taken of the length of the shaft, yet the torsional stiffness *is* affected by the length, in that it varies in inverse ratio to the length of the shaft. Combining the two rules, we say that the stiffness of a shaft varies as—

$$D^4 \div L, \text{ or } \frac{D^4}{L}$$

D being the diameter and L the length of the shaft. Thus a shaft 3 in. diameter and 100 ft. long is of about equal stiffness to a shaft 2 in. diameter and 20 ft. long.

Another very important point that must never be overlooked in calculating the diameter of any required shaft is the amount of the transverse load or stress which will come upon the shaft between the bearings, due to the weight of the wheels or pulleys fixed on the shaft, and also the tension of the several straps or belts. In this respect the shaft must be considered as a circular beam or girder, and if its diameter, as calculated in this manner, is greater than the diameter as fixed by consideration of the torsional strength and stiffness only, then such greater diameter must, of course, be provided.

Wrought-iron Girders.—Wrought iron, although not so readily constructed into the required form or shape, is a much more reliable material for girder and bridge building than cast iron. The tensile and compressive strengths of the metal are practically the same, and, moreover, wrought iron has the advantage of the quality of toughness. Colloquially speaking, we say that "it bends before it breaks," and therefore a structure of wrought iron will seldom fail without giving warning that it is being subjected to a greater stress than it can safely sustain. Owing to the higher tensile strength of the metal, a wrought-iron girder will be much lighter than one of cast iron of equal strength; and as in commercial transactions both are sold by weight,

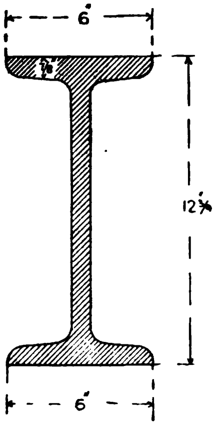


FIG 17

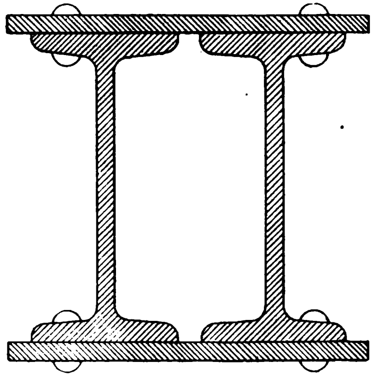


FIG. 18

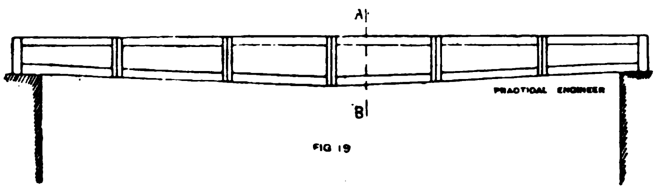


FIG 19

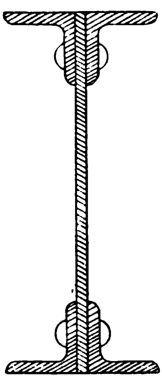


FIG 20

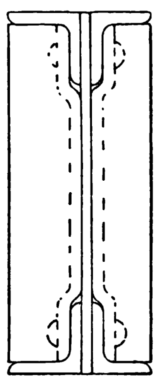


FIG. 21

a wrought-iron girder will generally be of less cost than a cast-iron girder constructed to carry a similar load.

The most common form of girder is that known as the "rolled-iron joist," so largely used in building constructions. Such joists or girders are produced by rolling the hot metal through suitably-shaped rolls, and hence, the amount of labour required being very moderate, they are produced at a small cost. The stresses coming upon a girder under different methods of loading and supporting have been already discussed when treating of cast-iron girders, and precisely the same principle applies to girders of wrought iron, or, indeed, of any other metal or material.

The following table gives reliable formulæ for the breaking strengths of *solid* rectangular girders of good wrought iron, under the different forms of loading and supporting illustrated in our first article :—

Girder (cantilever) supported at one end, loaded at the other—

$$W = \frac{1\frac{3}{4}\,B\,D^2}{4\,L}.$$

Girder (cantilever) supported at one end, load distributed—

$$W = \frac{1\frac{3}{4}\,B\,D^2}{2\,L}.$$

Girder supported at each end, loaded at centre—

$$W = \frac{1\frac{3}{4}\,B\,D^2}{L}.$$

Girder supported at each end, load distributed—

$$W = \frac{3\frac{1}{2}\,B\,D^2}{L}.$$

The safe distributed loads which can be carried by rolled-iron joists or girders, supported at each end, are generally published in the makers' or merchants' lists. As, however, a large proportion of the joists put upon the market are of continental manufacture, and the quality of the iron employed not easily ascertainable, it is advisable in all cases to check the published table of strength by calculating the maximum stress coming upon the girder, and allowing but a very moderate stress per square inch of section. Many engineers and architects insist on the employment of English-made girders in the contracts under their care, as they are then able to obtain definite information as to the quality of material employed.

As an example of the calculation of the strength of a rolled-iron joist or girder, we will take the case of a girder having a total depth of $12\frac{3}{4}$ in., with flanges 6 in. wide and $\frac{7}{8}$ in. mean thickness. It rests on supports at each end, the span being 16 ft.; we are required to find the safe permanent distributed load.

An end view of the girder is shown at fig. 17. As the load is to be evenly distributed throughout the span, the effect on the flanges at the centre of the girder, where the maximum stresses occur, will be precisely the same as the stresses produced by half the load to be carried, but placed on the girder at the centre of the span, midway between the two supports. The simplest method of arriving at the required result will be to ascertain the calculated breaking weight of the girder under the given condition of loading, and from that amount to fix the safe load by the factor of safety determined upon. The formula for the distributed breaking load will be as follows :—

Breaking load in tons = 4 (area of one flange × 20 × mean depth of girder ÷ half span).

Substituting the given dimensions in the above formula, we have—

$$\text{Breaking load in tons} = 4\left(6 \times \tfrac{7}{8} \times 20 \times 1 \div 8\right)$$
$$= 54\tfrac{1}{2} \text{ tons.}$$

It will be seen that we have taken the ultimate strength of the iron in tension and compression to be 20 tons per square inch. As the load is permanent, a factor of safety of 4 might be considered sufficient, but it will be better to adopt 5 as the factor. Hence the safe permanent distributed load on the girder is as follows, viz.—

$$54\tfrac{1}{2} \div 5 = 11 \text{ tons (nearly).}$$

To give increased strength, rolled-iron joists are frequently built up in pairs, by riveting cover plates across the flanges, in the manner shown in fig. 18. The effective area of each flange is found by taking the combined area of the two flanges and the cover plate, deducting the diameter of the rivet holes, because such holes weaken the tension flange.

In fig. 19 we illustrate a wrought-iron girder constructed of plates and angle bars, a type largely employed for carrying heavy loads across spans too great to be bridged by rolled-iron joists. An enlarged sectional view of the girder across A B is shown at fig. 20; whilst fig. 21 represents an end view, showing the method of finishing off the ends by short

angle bars. The method of calculation of strength is precisely similar to that employed with the rolled girder. The only point to note is that in taking the area of the flanges only the cross areas of the two angle bars which form each flange are considered. No account is taken of the portion of the central web plate between the angles, as such area compensates for the loss occasioned by the rivet holes.

CHAPTER IV.

Steel.

The employment of steel for the manufacture of edge tools of all descriptions, and for other kindred purposes, dates from a remote period. The steel so employed was produced almost entirely by the cementation process, in which bars of excellent quality wrought iron are converted into steel by heating them, in a closed furnace, in contact with carbon in the form of powdered charcoal. Such a process is still adopted for the purpose of tool steel, and the conical cementation furnace is conspicuous throughout the whole town and district of Sheffield, the seat of the steel trade.

But in recent years, and especially during the last fifteen years, the application of steel has been very much widened, so that at the present time we find it entering largely into constructive engineering works of all kinds. In many instances it has almost entirely displaced wrought iron on structures for which that material was formerly exclusively employed. A most notable instance is the Forth Bridge, constructed entirely of mild steel. This great advance in the application of steel has been brought about very largely by the cheapened methods of production associated with the names of Bessemer, Siemens, and others, who have introduced and made a commercial success of qualities of steel suitable, both in quality and cost of production, for constructional work.

To the engineer the information that a certain article or structure is made of *steel* affords but a very vague description, as it may indicate a material having a tensile strength of as much as 70 tons per square inch and upwards, or it may have a strength of but little, if anything, in excess of the strength of wrought iron. Moreover, such marked difference in the physical properties of the metal are brought about by very minute differences in the chemical composition. The

manufacture of steel requires, therefore, at all times the utmost care at every stage of its production, and during subsequent working in the building up of the desired structure. We may here refer to the difference of treatment required in the working of steel as compared with the working of wrought iron. A wrought-iron plate, for instance, may be bent or flanged piecemeal by ordinary smithing operations, the heating been accomplished in a smith's open-hearth fire. With a steel plate, however, it is necessary, if internal flaws are to be avoided, that the whole surface of the plate required to be flanged should be heated, and the flanging or bending accomplished as far as possible at one heat and in one operation by suitable machinery. The failures of some of the first constructed steel boilers were in a great measure due to the adoption of the same methods of working as those employed during construction of iron boilers.

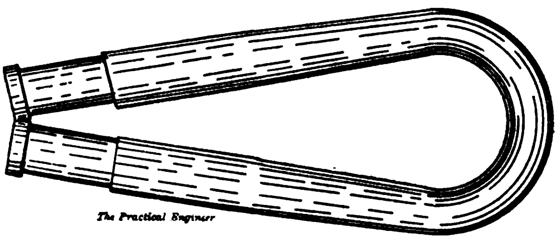


FIG. 22.

The difference in structure of wrought iron and steel has been not inaptly compared to the difference existing between a coarse and a fine textured fabric. A rent at the edge of a piece of coarse fabric would be in less danger of extending throughout the entire length or breadth of the material, than would a similar rent in a fine-textured fabric, although, taken as a whole piece, quite free from any rent, the fine fabric might resist a much greater stress than the coarse material. The structure of steel being much finer and of greater homogeneity than wrought iron, the illustration may serve to demonstrate the importance of guarding against initial flaws or defects of any kind.

For constructional purposes mild steel is chiefly employed. The term *mild* is used to distinguish from the harder qualities of crucible cast steel. The cheapest method of producing mild steel is probably the Bessemer process,

which is a method of obtaining steel direct from cast iron without previous conversion into wrought iron, as in the cementation process. The essential chemical process in producing steel in the cementation furnace is to impart or to put carbon *into* the metal, whereas in making steel direct from cast iron we require to take carbon *out* of the metal. In the early days of the Bessemer process it was the intention to only partially decarbonise the iron, until it had acquired the malleability and other properties of steel. For this purpose the molten cast iron was run into the Bessemer furnace or converter, where a strong current of air was blown through the metal to enable the oxygen of the air to

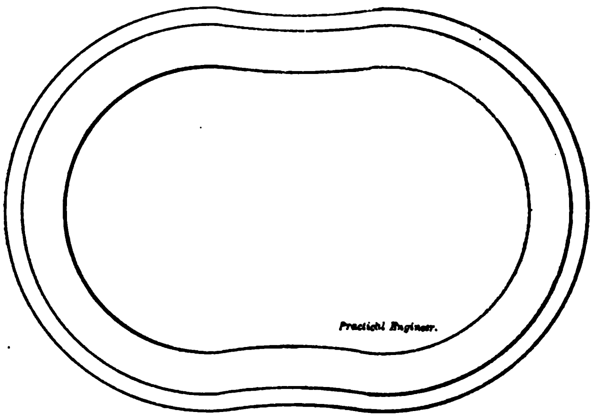


FIG. 23.

combine with and carry away a portion of the carbon. When only the required quantity of carbon remained in the metal the blast was shut off, and the charge run out from the converter into ingots, to be worked or rolled into rails, bars, or other forms required. It was found exceedingly difficult, however, to accurately judge as to when sufficient carbon had been removed, and the process was therefore modified, and the practice now is to blow air through the molten metal until the whole of the carbon has been removed, afterwards imparting the required quantity by throwing in a definite amount of spiegeleisen, a form of white pig iron very rich in carbon. We have given but the

briefest possible outline of the principles of the process; full details will be found in any work on metallurgy.

The tensile strength of Bessemer steel varies considerably, and is generally much increased by hammering and rolling. The metal, which in the form of the cast ingot has a tensile strength of about 26 tons per square inch, may be increased in tenacity to over 60 tons per square inch by hammering and rolling. Specifications for Bessemer steel axles usually stipulate that the steel shall have an ultimate tensile strength not less than 24 tons nor more than 32 tons per square inch, with not less than 20 per cent extension or elongation in a length of 8 in., and not less than 35 per cent contraction of area at point of fracture. An actual test gave the following results:—

Breaking load = 29·25 tons per square inch.
Elongation in 8 in. length = 24·2 per cent.
Contraction of area = 42·0 per cent.

Mild steel is also produced by, amongst others, the Siemens-Martin process, in which pig iron is melted in a furnace in contact with wrought iron and steel scrap, and also a quantity of rich oxide of iron and spiegeleisen.

Mild steel axles are frequently made with a tensile strength as high as 35 to 38 tons per square inch, and yet they are so tough that they may be bent double, as shown in fig. 22. The following are results of tests on five axles of this quality:—

No.	Breaking load. Tons per square inch.	Elongation in 2 in. length. Per cent.	Contraction of area. Per cent.
1	36	28	56
2	35	27	53·6
3	36	21·5	47·2
4	37·2	27·5	48·8
5	35	26	52·4

Steel boiler plates should have an ultimate tensile strength of 26 to 30 tons per square inch, the figures given being the minimum and maximum respectively, and should give an elongation of 20 per cent on a 10 in. test piece. A specification frequently reads as follows: Boiler plates to be made of Siemens-Martin steel of not less than 26 tons nor more than 30 tons ultimate tensile strength, with not less than 20 per cent extension in 10 in., nor less than 30 per cent contraction of area at fracture. Other engineers attach but

little importance to the amount of contraction of fracture, but are satisfied by taking careful notice as to the extension or elongation.

Railway tyres, having to withstand much wear on the tread or the rolling surface, are usually produced from a quality of crucible cast steel having a tensile strength of 45 to 50 tons per square inch, with an elongation of about 17 to 20 per cent and not less than 30 per cent of contraction of area. The tyres are forged and rolled from solid discs or blooms of the metal, and thus form, when completed, one solid ring of steel. One tyre representing each charge of metal taken from the furnace should be tested to destruction

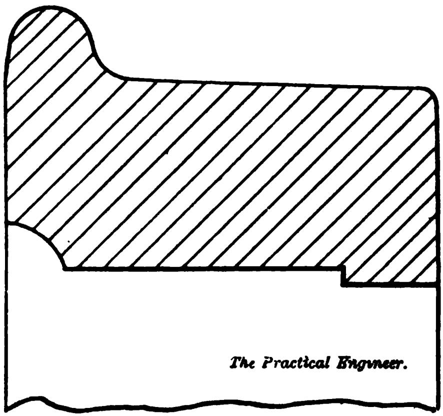


FIG. 24.

by allowing a weight or mass of iron to fall upon it, when standing in position on an iron block or plate, beneath the upright guides between which the test weight is raised and afterwards allowed to fall freely. Before fracture, the tyre should deflect from about $\frac{1}{8}$ to $\frac{1}{4}$ of its internal diameter, the amount depending upon the size of the tyre and also on the quality of the metal. The deflected tyre, after casting, is shown at fig. 23, and a section at fig. 24.

In ordinary steel castings, requiring no subsequent forging or rolling of any kind, the metal employed, although showing considerable tensile strength, will be very deficient in ductility, as indicated by the amount

of extension and contraction of area obtained by testing. The following results were obtained on a test piece taken from the steel employed in casting a number of trolley wheels :—

Breaking load	37·1 tons per square inch.
Elongation in 2 in. length......	9 per cent.
Contraction of area at fracture	7·4 per cent.

As indicating the minute difference in the chemical composition of qualities of steel having a marked difference in physical properties, we will now give two analyses, the first of a steel suitable for railway tyres, having a tensile strength of 48 tons per square inch, and the second of a steel employed for elliptical springs for railway wagons, having a tensile stress of 53 tons per square inch.

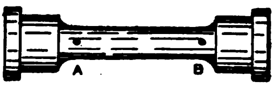


Fig. 25.

Fig. 26.

First analysis :—

Carbon.......................................	·62 per cent.
Manganese	·486 ,,
Phosphorus	·024 ,,
Sulphur	·028 ,,
Silicon......................................	·272 ,,

Second analysis :—

Carbon.......................................	·63 per cent.
Manganese	·438 ,,
Phosphorus	·040 ,,
Sulphur	·053 ,,
Silicon......................................	·032 ,,

In fig. 25 we illustrate the usual form of tensile test piece. The test length is measured between two centre punch dots *a*, *b*. The appearance of the fracture in the testing machine is shown at fig. 26. The elongation is measured by carefully holding the two pieces together, and then accurately scaling the distance by dividers and rule between the two centre dots.

We are now in a position to consider the increase of strength gained by the substitution of mild steel for wrought-iron shafting. The following results were obtained

by tensile tests on samples cut from $4\frac{1}{8}$ in. and $4\frac{5}{8}$ in. diameter ordinary mild-steel bars, intended for 4 in. and $4\frac{1}{2}$ in. shafting respectively :—

No. of test piece.	Breaking load.	Elongation in 10 in. length.
1	27·9 tons per square inch	24·5 per cent.
2	27·6 tons per square inch	24·5 per cent.

Taking wrought-iron shafting as having a strength of 20 tons per square inch, it will be seen that steel shafting has an advantage of one-third additional strength, so that the load required at the end of a 1 ft. lever to twist asunder a 1 in. diameter steel shaft of the quality indicated by given tests will be as follows :—

$$700 + \frac{700}{3} = 700 + 233$$

$$= 933 \text{ lb.}$$

In the same manner, other things being equal, a boiler constructed of steel plates will carry about 33 per cent greater pressure than a similar boiler constructed of iron plates of the same dimensions ; or, to carry the same pressures, plates of steel would be proportionately thinner than iron boiler plates. It would be practically impossible to construct many modern boilers with iron plates, as the high working steam pressures now used would necessitate the employment of plates too thick to be successfully jointed at the riveted seams.

CHAPTER V.

Strength of Spur or Toothed Gearing.

Before proceeding to treat of copper, brass, and other alloys as employed by engineers, we must consider a few of the important machine and constructive details which may be produced from either cast or wrought iron, or cast or forged steel.

Cast iron is the material chiefly employed for spur wheels. For small wheels of a solid or disc form wrought iron or forged steel may be employed, and in recent years cast-steel wheels of large diameter have met with much success. Such wheels may be of one casting, entirely of steel, or the teeth and rim portion only can be formed of that metal, built up in segments it may be, and secured

with bolts and nuts, cottars and wedges, to each other, and to the cast iron or steel arms connecting to the central hub or boss of the wheel.

But of whatever metal a spur wheel is constructed, the strength of its teeth is governed by the same general principles. Each tooth of the wheel constitutes a cantilever beam or girder, and its power of resistance to the load imposed upon it must be calculated accordingly. The root or end of the tooth adjacent to the rim is, of course, the

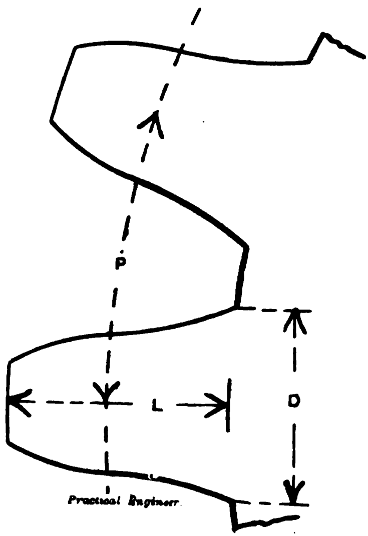


Fig. 27.

fixed end of the cantilever, and the extreme end or face of the tooth is the free end. We will now proceed to take a practical example.

Fig. 27 represents two wheel teeth of a strong form, the pitch being, say, 2 in. The other dimensions may be taken as follows :=

Length of tooth, $L = \frac{3}{4}$ P (pitch) $= 1\frac{1}{2}$ in. $= \frac{1}{8}$ foot.
Thickness of root, $D = \frac{2}{3}$ P $= 1\frac{1}{3}$ in.
Width or breadth of face = 5 in.

Now, in our first article we have seen that the breaking strength of a solid rectangular cast-iron cantilever is expressed by the formula—

$$\text{B.W.} = \frac{\text{B D}^2}{4\text{ L}}.$$

Where B.W. = breaking load in tons on end of girder
B = breadth of girder in inches
D = depth of girder in inches
L = length of girder in feet.

Therefore, by employing this formula with the given dimensions of the tooth, we obtain—

$$\text{Breaking load at end of tooth} = \frac{5 \times (1\tfrac{1}{3})^2}{4 \times \tfrac{1}{8}}$$

$$= \frac{5 \times \tfrac{16}{9}}{\tfrac{1}{2}}$$

$$= \frac{5}{1} \times \frac{16}{9} \times \frac{2}{1}$$

$$= \frac{160}{9} = 17{\cdot}78 \text{ tons.}$$

By employing a factor of safety of 10, the safe load at the end of the tooth becomes 1·778 ton, or about 3,920 lb., and we see that if the wheel teeth were but 1 in. pitch, the safe load would be 392 lb. Other things being equal, then, it may be taken that the strength of wheel teeth increases directly as their pitch, and as the width or breadth of their faces.

A rule for the strength of cast-iron wheel teeth, as given by Box in his treatise on "Mill Gearing," is as follows :—

$$\text{S.L.} = \text{P} \times \text{W} \times 350$$

where S.L. = safe load on 1 tooth in pounds
P = pitch in inches
W = width of face in inches.

It will be observed that this is a very safe rule, allowing for the fact that the metal in the teeth may not be equal to that in a test bar taken from the same material, in addition to providing a factor of safety of 10.

When two wheels of different diameters are geared together, the smaller of the two is frequently termed the pinion. In general it may be assumed that two teeth of

each wheel will be in gear together; but when very small pinions are used, such as those employed for cranes and other lifting machinery, only one tooth may be engaged at a time. The same effect is produced if the teeth are not pitched with the greatest possible care and accuracy, or if they do not all wear at precisely the same rate. As these conditions cannot be absolutely ensured, we must generally consider that the whole load to be transmitted from wheel to wheel has to be sustained by one tooth.

In setting a pair of wheels in gear, great care must be taken to ensure that the teeth shall properly bear upon each other across the whole width of the face, as otherwise the load will not be properly distributed, and the teeth will therefore be unable to put forth their full strength.

With teeth cut in the solid metal on a wheel cast with a solid rim, it is generally easier to obtain the proper adjustment of the gearing than with wheels cast in the ordinary manner. In fig. 28 we have a plan of the top of a wheel tooth having its two sides perfectly parallel. With cast

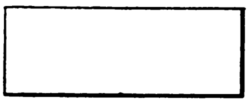


FIG. 28.

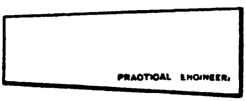


FIG. 29.

teeth, however, the sides will not be perfectly parallel, because a slight taper must be made on the pattern to enable it to be properly withdrawn from the mould, and the teeth of the wheel itself will therefore be cast with slightly tapered sides, as shown in an exaggerated form in fig. 29. It is evident that to ensure equal bearing along the entire width of such teeth they must be set into gear in the manner shown in fig. 30—that is to say, with the wheels so fixed upon their shafts or axles as to cause the wide sides of the teeth to come opposite to each other on the respective wheels, and not adjacent, as shown in fig. 31. The shaded tooth in each figure belongs to one wheel, and the unshaded teeth to the other wheel of the pair in gear.

In the design of toothed gearing, in addition to the strength of the teeth, we have to consider the proper form to be given to them, so that in working there shall be but little noise, and small loss from friction. The motion of a pair of spur wheels should be as smooth and regular as the rolling of two plain friction wheels or rollers upon each other, having diameters equal to the pitch diameters of the

spur wheels. Such conditions can be fulfilled by teeth having considerable variation in their proportions, and the strength of the tooth will, of course, be affected by its form, inasmuch as its dimensions are varied. The teeth shown in fig. 27 are especially strong, because the flanks or portions below the pitch line are so shaped as to give a good depth or thickness of metal at the root. Teeth of small wheels or pinions, and also the teeth of racks, are frequently very weak at the roots, and in all wheel teeth having radial flanks the depth or thickness at the root will be less than the thickness at the pitch line; such teeth are therefore deficient in strength.

To strengthen up weak teeth the practice of shrouding is frequently adopted, which consists in extending the rim of the wheel and carrying a flange of metal up each side of the

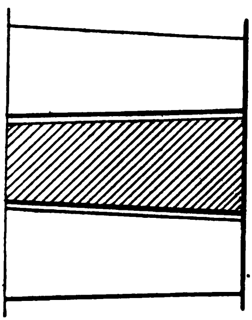


FIG. 30.

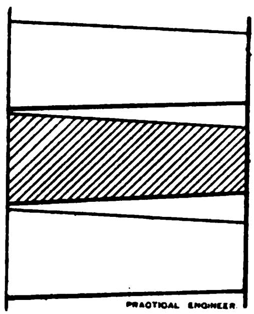


FIG. 31.

teeth, as shown by the part sectional view at fig. 32. When a wheel is made with shrouding, as in fig. 32, it is said to be shrouded up to the pitch line; sometimes the shrouding or flanging is carried to the top of the teeth, but then, of course, only one wheel of a pair that are to be geared together can be shrouded, whereas both wheels of a pair may be shrouded almost up to the pitch line, and thus both will be strengthened. The effect of shrouding is to reduce the length of the cantilever, for in a wheel shrouded up to the pitch line the length of the overhanging tooth will be reduced from the full depth of the tooth to the depth to the pitch line only, and thus, as the length of cantilever is now less than half its former length, the strength of the tooth should be more than doubled. No purpose is served by

shrouding to the pitch line a wheel having teeth of the form shown in fig. 27, because, owing to the parabolical shape of the flanks, the teeth are as strong at the roots as at any other part, and there will be little or no tendency for the tooth to break away at any one part before another. But if such teeth are shrouded to the top, their strength will be nearly doubled, though, as we have previously indicated, such addition of strength can be obtained on one wheel only of a pair, and is thus of little advantage.

A shrouded wheel has a somewhat heavier appearance, but it is a ready and effective method of strengthening up teeth of a weak form. With parallel-flanked teeth, shrouding to the pitch line is generally considered to increase the strength one-third, whilst the addition in the case of radially-flanged teeth may be taken at as much as 100 per cent. In each case the reference is to shrouding on each side of the

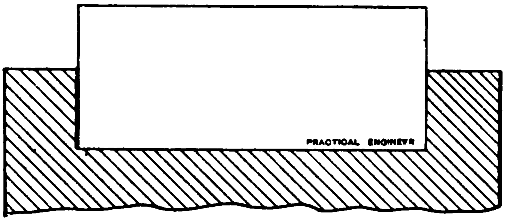


FIG. 32.

wheel, or double shrouding, as it is frequently termed, in the manner indicated in fig. 32. Want of space will sometimes forbid double shrouding, and single shrouding (or with the flange on one side only) is the only means of strengthening. The extra strength so obtained will be much less than one-half that obtained with the double shroud.

In the foregoing consideration of the strength of wheel teeth we have considered the load applied at the end of the tooth to be a static or dead load. The strength of gearing for cranes and other slow-moving machinery may be calculated by the rules already given, but with quick-moving mill gearing another principle must be considered. The actual load or force transmitted in mill gearing is usually very small compared with the force transmitted through crane and similar gearing. With mill gearing, however, the teeth have to resist a load suddenly brought to bear upon them during the rapid revolution of the wheels, and hence

such teeth have to resist considerable shocks or impacts. The strength of a solid girder to resist impact may be considered to depend directly on the quantity or weight of metal in it, and it will thus vary directly as the length, thickness, or depth, and width of the girder. With wheel teeth the length and depth increases in direct proportion with the pitch; thus the teeth of a wheel 4 in. pitch will be twice as long and twice the thickness or depth of the teeth of a wheel 2 in. pitch. The strength of teeth to resist impact, therefore, varies as the pitch squared. Based on this principle, several formulæ have been published for ascertaining the power that may be transmitted by mill gearing. The following simple formula, by Hutton, expresses the capability of ordinary cast-iron gearing :—

$$H.P. = \frac{P^2 \times W \times D \times N}{240}$$

where P = pitch in inches
W = width of face in inches
D = diameter to pitch line in feet
N = number of revolutions per minute
240 is a constant number established by experiment and experience
H.P = actual horse power that can be safely transmitted.

Thus, for example, to find the horse power that can be transmitted by a wheel of 4 ft. pitch diameter, 3 in. pitch, 7 in. face, running at 40 revolutions per minute, we proceed as follows :—

$$\begin{aligned} H.P. &= \frac{9 \times 7 \times 4 \times 40}{240} \\ &= 3 \times 7 \times 2 \\ &= 42. \end{aligned}$$

It will be noted that, as in the case of shafting, the power that can be transmitted by mill gearing varies directly as its speed.

CHAPTER VI.

Spur or Toothed Wheels.—*Continued.*

Although the formula given in our last article may be taken as generally accurate for ordinary cases of power transmission by cast-iron spur wheels, with ample margin for safety, yet it must not be considered as applicable to every case without modification. With very slow running

gearing, where the load on the teeth may be considered as a static load, we can, as already shown, very accurately calculate the load that may be put upon the teeth. But with a quick-running gearing, every case should really be treated separately, regard being taken to the pitch line velocity of the wheel, as well as the actual load or pressure on the teeth and the resistance to impact, calculated from such data.

A consideration that should not be overlooked is the effect of a high velocity on the rim of a spur wheel, for if the pitch line speed becomes excessive, the rim will burst in consequence of the centrifugal tension, after the same manner that a flywheel rim occasionally bursts. It might appear that an increase of the section of the rim would overcome this difficulty, but it will not do so, because the centrifugal tension increases in the same ratio as any increase in the weight of the rim; and hence, for very high velocities, steel or some other material stronger than cast iron must be employed. Professor Unwin shows that the limit of a safe velocity of the rim of a tooth wheel must be taken at about 96 ft. per second as a maximum, and that heavy wheels must be kept well under this speed. In ordinary mill work, however, the necessity for quiet and steady running of gearing will cause the velocity to be kept very much below the maximum limit. A rim speed of 96 ft. per second corresponds to a spur wheel, 10 ft. pitch diameter, running at about 200 revolutions per minute.

The relative strengths of wheel teeth under a static or dead load may be considered approximately as follow:

Cast iron	1
Malleable cast iron	2
Wrought iron..	2 to 3
Cast steel..	3 to 4
Gun metal	2

These are fairly average values. By obtaining definite tests and working data as to the quality of the particular metal to be employed, the true relative value can be readily assigned.

In mill gearing, transmitting power when running at velocities up to 3,000 ft. per minute (which may be considered as the safe maximum pitch velocity), Messrs. Musgrave consider that the relative capacities of two similar wheels, one of cast iron, the other of cast steel, is in the ratio of 625 to 1,000, or approximately 5 : 8. Thus, if a certain cast-iron spur wheel, running at a given speed,

can transmit 10 horse power, then a similar wheel in cast steel, running at the same speed, will be capable of transmitting 16 horse power.

In the design and arrangement of gearing it should be remembered the finer the pitch the more smooth and uniform will be the motion of the wheels, and hence the pitch should always be made as small as possible consistent with the necessary strength. For heavy work, such as the driving of heavy rolling mills and planing machines, where coarse pitched wheels must of necessity be employed, the teeth are sometimes "stepped," in order to obtain a more smooth and uniform motion than would be otherwise possible. Such stepping is illustrated by the plan of a

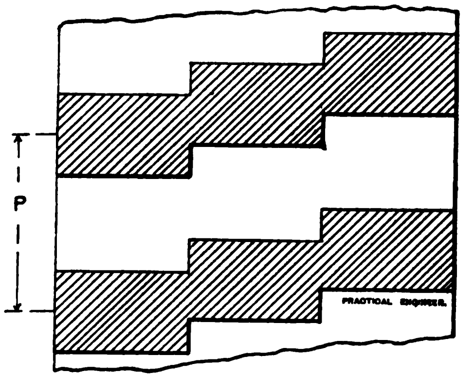


FIG. 33.

portion of a wheel in fig. 33. The pitch of the teeth is the distance P, as in ordinary teeth, but in the step teeth, as illustrated, there are three steps instead of one between the teeth, the effect of the arrangement being that we approximately obtain the same smoothness of motion as with teeth having only one-third the pitch, whilst at the same time the additional strength due to the larger pitch is obtained.

But a much more effective method of obtaining smooth motion with coarse pitched teeth is by the adoption of helical gear, a form of wheel teeth now very largely employed. An illustration is given at fig. 33A, showing plan of double helical teeth. With such a wheel we not only obtain a smooth motion, but also increase of strength due to

the extra effective width of the teeth; at the same time there is no end pressure on the shaft, such as would be occasioned with single helical teeth, because with the double helical form there is an equal tendency to end pressure in each direction, and thus the two opposite forces balance each other.

CHAPTER VII.

Iron and Steel Girders.

We have already considered at some length the general principles governing the strength of solid rectangular girders. Such principles apply to every beam or girder of whatever material it may be constructed; the results obtained will therefore depend on the strength and properties of the particular material employed. The

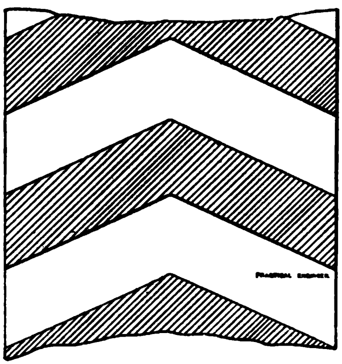


Fig. 33a.

following table gives the breaking loads of solid beams or girders of different materials under varying conditions of loading and supporting. The cast-iron and wrought-iron values have been previously given, but we repeat them here to make the table complete. It should be noted that in a girder, supported firmly at each end by bolting down, or building in, or otherwise, instead of simply resting upon its supports, considerable addition of strength is obtained over the values given in table; the increase in some instances may amount to 50 or even 100 per cent.

It is usual to make the depth of the girder one-twelfth of span when supported at both ends, or one-sixteenth of

TABLE OF BREAKING WEIGHT IN TONS OF SOLID BEAMS OR GIRDERS.

L = length or distance between supports in feet.
B = breadth in inches.
D = depth in inches (or diameter in circular section).

	Cast iron.	Wrought iron.	Steel.	Oak.
CANTILEVERS, LOADED AT ONE END, SUPPORTED AT THE OTHER END.				
Rectangular section	$\frac{B D^2}{4 L}$	$\frac{1\frac{3}{4} B D^2}{4 L}$	$\frac{3 \text{ to } 5 B D^2}{4 L}$	$\frac{\frac{1}{7} \text{ to } \frac{1}{4} B D^2}{4 L}$
Circular section	$\frac{\frac{9}{16} D^3}{4 L}$	$\frac{1\frac{1}{8} D^3}{4 L}$	$\frac{2 \text{ to } 3\frac{1}{4} D^3}{4 L}$	$\frac{\frac{1}{12} \text{ to } \frac{1}{7} D^3}{4 L}$
CANTILEVERS, WITH DISTRIBUTED LOAD.				
Rectangular section	$\frac{B D^2}{2 L}$	$\frac{1\frac{3}{4} B D^2}{2 L}$	$\frac{3 \text{ to } 5 B D^2}{2 L}$	$\frac{\frac{1}{7} \text{ to } \frac{1}{4} B D^2}{2 L}$
Circular section	$\frac{\frac{9}{16} D^3}{2 L}$	$\frac{1\frac{1}{8} D^3}{2 L}$	$\frac{2 \text{ to } 3\frac{1}{4} D^3}{2 L}$	$\frac{\frac{1}{12} \text{ to } \frac{1}{7} D^3}{2 L}$
GIRDERS, SUPPORTED AT EACH END, WITH LOAD AT CENTRE.				
Rectangular section	$\frac{B D^2}{L}$	$\frac{1\frac{3}{4} B D^2}{L}$	$\frac{3 \text{ to } 5 B D^2}{L}$	$\frac{\frac{1}{7} \text{ to } \frac{1}{4} B D^2}{L}$
Circular section	$\frac{\frac{9}{16} D^3}{L}$	$\frac{1\frac{1}{8} D^3}{L}$	$\frac{2 \text{ to } 3\frac{1}{4} D^3}{L}$	$\frac{\frac{1}{12} \text{ to } \frac{1}{7} D^3}{L}$
GIRDERS, SUPPORTED AT EACH END, WITH LOAD DISTRIBUTED.				
Rectangular section	$\frac{2 B D^2}{L}$	$\frac{3\frac{1}{2} B D^2}{L}$	$\frac{6 \text{ to } 10 B D^2}{L}$	$\frac{\frac{2}{7} \text{ to } \frac{1}{2} B D^2}{L}$
Circular section	$\frac{1\frac{1}{8} D^3}{L}$	$\frac{2\frac{1}{4} D^3}{L}$	$\frac{4 \text{ to } 6\frac{1}{2} D^3}{L}$	$\frac{\frac{1}{6} \text{ to } \frac{2}{7} D^3}{L}$

length in the case of a cantilever, though such proportions are very much altered to suit particular requirements.

Rolled Steel Joists.—These are now largely made by English makers, and generally may be obtained, strength

for strength, at a lower cost price than rolled-iron joists. The method of calculating the strength is precisely the same as with an iron joist, but as the tensile and compressive strength of the mild steel employed is higher than that of the iron, a given section steel joist will support a greater load than a similar section in iron. Most makers of steel joists keep to the well-known stock sections, such as the 8 in. by 4 in., 10 in. by 5 in., 12 in. by 6 in., &c. (the figures expressing the over-all depth and width of flanges respectively), but make them of less thickness of metal than

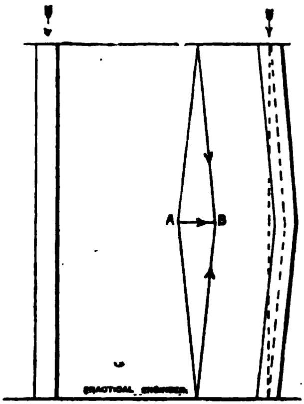


FIG. 34. FIG. 36. FIG. 35.

the iron joists; and hence a foot run of steel joist will be lighter than an equal length in iron; an important advantage, effecting, in many instances, considerable saving of time and labour in erection.

Pillars or Columns.

If no deflection occurred in a loaded pillar or column, then its strength would be simply measured by its compressive resistance. But in every column of considerable length, any load placed upon it will cause deflection, and such deflection entirely alters the resisting power of the structure. In the case of a round column of cast iron, for instance, if it could be maintained perfectly vertical under its load, then the metal, being simply in compression, would be as suitable as any that could be

employed for a steady load, because of the great compressive strength of cast iron. But when deflection takes place, and the column, instead of being truly vertical, as in fig. 34, is forced by the load into a position similar or approaching to that shown in fig. 35, then it will be readily seen that in addition to the vertical load there is also a transverse load acting on the column, which can be measured by the length of the diagonal A B in the parallelogram of forces at fig. 36. Many careful experiments and investigations have been made in connection with the strength of columns, and as a result the following laws, applicable to all columns or pillars of whatever material, have been established: (1) The strength of a column varies inversely as the square of its length. (2) The strength of a column is directly proportional to the fourth power of the diameter, or the side of the square in the case of a square column. Combining these rules, we get the formula, $D^4 \div L^2$, as expressing the proportional strength of columns. Thus a column 20 ft. high and 10 in. diameter will have but little more than one-half the strength of a column 10 ft. high and 8 in. diameter.

CHAPTER VIII.

Copper, Brass, Bronze, and Bearing Metals.

The application of pure, or commercially pure, copper in mechanical engineering and in manufacturing arts is very limited, but when alloyed with small quantities of other metals it is very largely employed for a great variety of purposes. In its purest commercial state, and very largely in the form of wire, an important application of copper is found in electrical engineering, its high conductive power rendering it especially suitable for the conveyance of electric currents. The best wire for such a purpose consists of as much as 99 per cent pure copper, whilst any wire containing less than 96 per cent copper is unfit for electrical work.

The physical properties of copper are well known. Its colour is red, it is tough and of considerable tenacity. The tensile strength of cast copper may be taken at 10 tons per square inch. The tenacity is considerably increased by proper working—by rolling or hammering; hence sheet copper or rolled bars will have a higher strength. Ordinary wrought-copper bolts have an ultimate tensile strength of about 15 tons per square inch. On account of its malleability

copper is very suitable for rolling into sheets, and pressing or hammering into thin hemispherical or other desired shapes. In the form of sheets it is employed for locomotive fireboxes, its heat-conducting power being of great advantage, but it should be noted in this connection that the tenacity of copper is affected by heat to a much greater extent than is the case with wrought iron or steel. Copper castings are unreliable; the metal does not run well in the mould, and the castings produced are porous. The addition of small quantities of phosphorus, up to about 4 per cent, has the effect of producing greater soundness in casting.

Brass.—The production of this well-known alloy absorbs a great quantity of copper. In Birmingham, the seat of the brass trade, it is estimated that there are about 35,000 persons, including men and women, boys and girls, employed in the various branches of the brass industry, and that the annual consumption of copper is about 34,000 tons. Excellent castings are produced with brass, and, being a good metal for working with cutting and burnishing tools, it can be readily got up to present a pleasing and ornamental appearance. The chemical composition of brass varies with the purpose of its application. Ordinary brass for castings should have a composition of 66 per cent copper to 33 per cent zinc, or say 2 of copper to 1 of zinc. Best brass for foundry purposes has 70 per cent copper to 30 per cent zinc; this is the composition employed for the tubes of locomotive and marine boilers. Muntz metal, or yellow metal as it is sometimes termed, contains about 60 per cent of copper to 40 per cent of zinc, or say 3 parts of copper to 2 parts of zinc. A little lead, about 1 per cent, is very frequently added to brass in order to make the metal soft and easy for tooling. Unlike copper, which may be rolled either hot or cold, brass can only be rolled cold, and in such rolling great care is required, the strip having to be frequently annealed. Muntz metal can be rolled either hot or cold. The tensile strength of brass varies from about 8 to 12 or 13 tons per square inch.

Bronze or Gun Metal.—This is a hard, tough, and strong alloy of copper and tin. Copper itself is soft, as also is tin, but by alloying the two metals we obtain hard bronze or gun metal. The tensile strength of this metal will vary with its composition, and also with the method of casting. Its average tensile strength may be taken at 14 tons per square inch. The composition of ordinary gun metal is 90 per cent copper and 10 per cent tin. The greater the

proportion of tin the harder the metal; hence for bearings of quick-running machinery a form of gun metal is employed having from 12 to 14 per cent of tin.

Phosphor Bronze.—This alloy of copper and tin, with a small percentage of phosphorus, is harder, and at the same time much stronger and tougher, than ordinary gun metal. It is used for large marine-engine bearings, also for gearing subjected to much shock. Its tensile strength varies from 22 to 32 tons per square inch.

Delta Metal.—The name applied to this metal is but a name only, and in no way indicates either its properties or the purpose to which it may be suitably applied. It is an

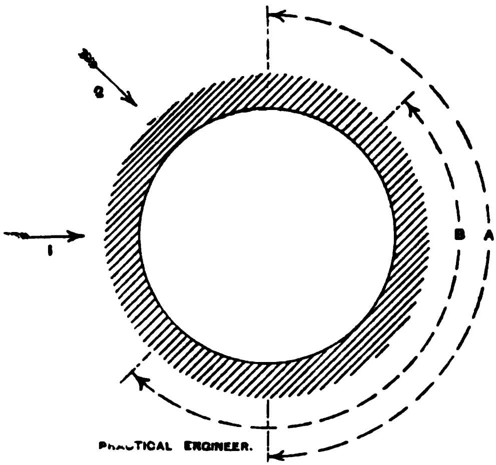


FIG. 37.

alloy of somewhat recent introduction. The chief constituents have been given as zinc alloyed with iron and with copper. It is claimed for delta metal that it may be rolled hot or cold, that it may be cast, also forged or stamped hot; it has but small liability to corrosion, and is not magnetic. Its tensile strength is very high according to the statement that has been given of its properties in this direction, as follows: Cast in sand, 20 to 24 tons per square inch; forged, 35 to 38 tons per square inch; whilst by hammering cold the tensile strength may reach 40 tons, and in the form of wire as much as 60 tons per square inch.

White Metals for Bearings.—There are many of these in the market, having in some instances very fanciful names, and with wonderful properties advanced concerning their suitability for shaft bearings. Some of them are fusible at very low temperatures, and may be cast in position round a smooth mandrel representing the shaft, thus effecting a saving of the boring or turning required for ordinary gun-metal or similar bearings, and also allowing of easy renewal when the metal wears. The composition of white bearing metals admits of wide variation, but the chief elements of the alloys are lead, tin, zinc, antimony, and copper. What is known as Babbitt's metal usually contains 96 parts tin, 8 parts antimony, and 4 parts copper.

Friction of Bearings.—The coefficient of friction between two metals in contact will depend upon the quality of the metal, the state of the rubbing surfaces as regards their smoothness and evenness, and also in respect to their lubrication. With good ordinary plummer blocks or bearings, having ordinary methods of lubrication attached to same, the coefficient of friction is usually taken at 0·7, or about $\frac{1}{14}$th of the full pressure of the bearing. Many of the white metals give a much lower coefficient.

It has been found by practical experience that when the pressure per square inch of rubbing surface of a bearing exceeds a certain limit, the oil or other lubricant is squeezed out, and a metallic contact occurs between the two rubbing surfaces, with the result that the shaft and bearing cut and abrade each other, and that such abrasion will take place in spite of constant lubrication. The tendency to abrasion is simply proportional to the pressure on the bearing, other things, such as lubrication, state of surfaces, &c., being the same. Experiments have shown that with a steel or wrought-iron shaft running in a cast-iron bearing abrasion will be likely to occur when the pressure exceeds from 5 cwt to 6 cwt. per square inch of rubbing surface; with a similar shaft, running in an ordinary brass or gun-metal bearing, the limiting pressure is 7 cwt. per square inch of rubbing surface.

The rubbing surfaces are the surfaces of the shaft and bearing in actual rubbing contact. Thus, in reference to fig. 37, it will be seen that when a load presses a shaft on to its bearing in the direction indicated by the arrow 1, the extent of actual rubbing surface is the semi-circumference A, multiplied by the length of the bearing or journal.

When the load comes in the direction shown by the arrow 2, then the semi-circumference B, multiplied by the length as before, will give the surfaces in contact. Thus, in a bearing having the brass or other liner or bush around the entire circumference of the shaft, the maximum load that can be carried by the brass-lined bearing is as follows :—

$$7 \times \text{semi-circumference of shaft} \times \text{length of bearing} = \text{load in cwts.}$$

But in practice we must keep well under this maximum load, and for constantly-running shafts the load should not exceed about 300 lb. per square inch of rubbing surfaces.

In figure 37 the circumferential length of the rubbing surface subtends an angle of 180 deg., but the length should be such as to subtend an angle not greater than 90 deg. Where the pressure may come in two opposite directions, then the brass will be in two segments, each of about one-quarter circumference; one segment will be usually at the top, the other at the bottom, the intervening spaces on either side being filled up with pads. In railway axle boxes the brass comes at the top only. Bearings having segmental brasses in this way have less than half the friction and require less oil than ordinary bearings having complete circumferential brasses.

A bearing will run hot from either of two causes (assuming that the workmanship is good and that the heating is not due to defects in fitting and erection): First, if the pressure is too great, and, secondly, if speed is excessive. We have therefore to fix a limit to $P \times V$, where P is the pressure per square inch of rubbing surfaces, and V the velocity of moving surfaces in feet per minute. Such a limit, with ordinary shafting and bearings with ordinary lubrication, must be taken at 20,000 as a maximum. Thus, suppose we have a 3 in. diameter shaft, running in an ordinary bearing, and that the pressure is, say, 300 lb. per square inch of rubbing surfaces. Then the speed, the periphery speed, of the shaft must not exceed $20{,}000 \div 300 = 66$ ft. per minute. In one revolution of the shaft the distance travelled by any point in the circumference $= 3 \times 3{\cdot}1416$, or, say, $\frac{3}{4}$ ft.; hence the maximum speed at which shaft may be driven $= 66 \div \frac{3}{4} = 88$ revolutions per minute.

CHAPTER IX.

TRANSMISSION OF POWER BY BELTING.

BELT gearing, consisting of drums or pulleys secured to a driving and driven shaft respectively, and connected by a flexible leather belt, is usually employed to transmit power through considerable distances in machine shops and factories, or where the conditions are favourable, and absolute precision of revolution of the driven shaft not essential.

The principle of such application of belting depends chiefly upon coil friction. When an ordinary hempen rope

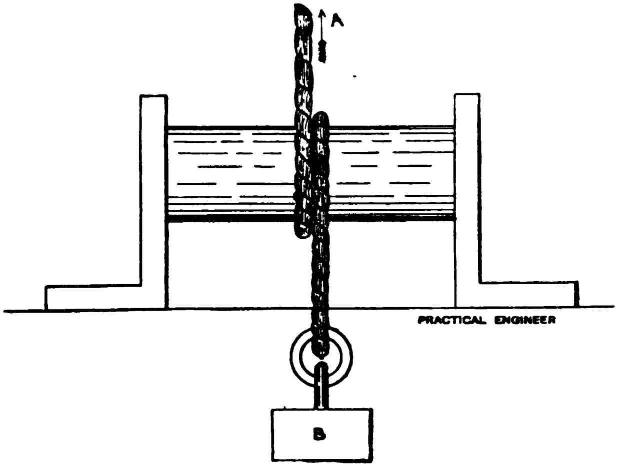


FIG. 38.

is coiled, say, once completely round a fixed wooden axle or drum, the friction between the coil and the axle is so great that a weight represented by 1 at the end of A (see fig. 38) will sustain a weight about nine times as great at the end B. If the rope be coiled twice round the barrel, then a weight of 1 at the end A will sustain a load of $9 \times 9 = 81$ at the end B; and if the coil is three times round, the difference between the weights will be $9 \times 9 \times 9 = 721$. The values of these ratios as given by various experimenters differ very widely, but the result will be much affected by the state of the surface of the fixed axle or drum; a rough

wooden axle or drum will give a greater ratio than a smooth surface of iron.

From the standard experiments on friction carried out by General Morin, we find that the coefficient of friction between a leather strap or belt and the surface of a pulley is as follows :—

Leather belts in ordinary working order on wooden pulleys = ·47.

Leather belts in ordinary working order on cast-iron pulleys = ·28.

Leather belts when soft and moist on cast-iron pulleys = ·38

In belt driving we are seldom able to embrace more than three-fourths of the whole circumference of the pulley, and usually not more than one-half. The ratios of the tension on the tight and slack sides of the belt will vary with the coefficient of friction; the higher the coefficient the greater the ratio. If the belt is soft and moist, and consequently very pliable, the higher coefficient given above may be relied upon, but the following table is based on a coefficient of ·28.

Proportion of circumference of pulley embraced by the belt.	Tension on tight side of belt when tension on slack side is represented by 1.
$\frac{1}{8}$	1·25
$\frac{1}{4}$	1·5
$\frac{3}{8}$	2
$\frac{1}{2}$	2·5
$\frac{5}{8}$	3
$\frac{3}{4}$	3·75

With two pulleys of unequal diameters, the large pulley will have more than half its circumference embraced by the belt, but on the smaller pulley the belt will be in surface contact for less than half the circumference. In such cases we must always be careful to make our calculations from the smaller pulley. With a crossed belt, as in fig. 39, the arc or proportion of circumference embraced by the belt will be the same for each pulley; but, owing to the extra wear and tear due to the rubbing of the belt on itself, a crossed belt should not be employed unless absolutely necessary.

The tension on the slack side is produced by the initial tension put upon the belt in stretching it over the two pulleys, and also, in favourable conditions, by the weight of

the slack side of the belt. It should be arranged, wherever possible, that the top side of the belt is the slack side, because the sag or drop of the belt will then cause a greater arc to be embraced on each pulley. Thus in fig. 40, where the bottom side is the slack side, it will be seen that the sag of the belt lessens the arc of contact on each pulley, whereas in fig. 41, having the slack on the top side, the arc of contact is increased. In each figure A is the driving and B the driven pulley.

The ratio between the slack and tight sides of a belt is independent of the breadth or width of the belt, but as a wider strap has greater strength than a narrower one, a greater tension can be sustained by it, and thus more power may be transmitted. The ratio of tensions is also independent of the diameters of the pulleys, though it is

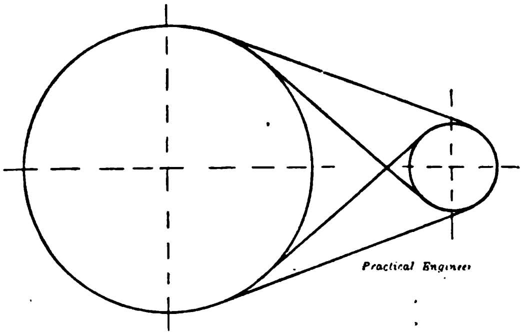


FIG. 39.

frequently assumed that the ratio will increase somewhat with an increase in diameter of the pulleys employed. It is quite true that a belt will run better on a pair of large pulleys, but the reason is that it has to undergo a less degree of bending and straining than when running over small pulleys. The greater the horizontal distance between the centres of two pulleys, the better the belt will run; it is very disadvantageous to have two pulleys directly vertical over and under each other, for the initial tension on the slack side has then to be produced entirely by the stretching of the belt on the pulleys, and it has thus always to be kept very tight.

The strength of a leather belt to resist a tensile stress has now to be considered. The ultimate tensile strength of

ordinary leather belting is about 3,500 lb. per square inch. The average thickness of single belting is $\frac{7}{32}$ in., and thus a 1 in. width of such belt has an ultimate strength of $3{,}500 \times \frac{7}{32} = 770$ lb. nearly. But the strength must be calculated from the weakest part, which will be the laced joint, having a strength of not more than one-third of the solid belt. Hence the breaking strength becomes $770 \div 3 = 256$ lb. per 1 in. width. The safe working strength is generally taken at 70 lb. per 1 in. width, or somewhat more than one-fourth the breaking stress. To assist the memory it is sometimes put in this way: The working strength of 1 in. width belting is equal to 20 lb. for each $\frac{1}{16}$ in. in thickness; thus a double belt $\frac{7}{16}$ in. thick has a strength of 140 lb. The strength

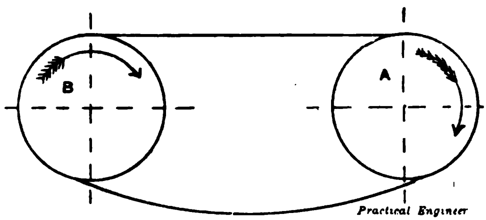


FIG. 40.

increases in direct proportion with the width, but a greater width than 18 in. is very seldom employed.

We will take the following question as an example of the transmission of power by belting:—

An engine running at 100 revolutions per minute gives 10 brake horse power. The whole of such power is transmitted from the crank shaft by a belt running upon a 5 ft. diameter pulley and a similar pulley on a countershaft. What width of belting $\frac{7}{32}$ in. thick will be required?

$$10 \text{ B.H.P.} = 10 \times 33000.$$
$$= 330000 \text{ foot-pounds per minute.}$$

The rim speed of pulley or speed of belt

$$= 5 \times \frac{22}{7} \times 100,$$
$$= 1571 \text{ ft. per minute.}$$

$\therefore$ tension or force at rim of pulley

$$= \frac{330000}{1571}$$
$$= 210 \text{ lb.}$$

Now, as the driving and the driven pulleys are of equal diameter, one-half the circumference of each pulley will be embraced by the belt, and therefore the ratio between the tension on the slack side of the belt, the force at pulley rim, and the tension on tight side of belt will be

$$1 : 1\tfrac{1}{2} : 2\tfrac{1}{2} = 2 : 3 : 5$$

$$\therefore \text{ maximum tension on belt} = 210 \times \tfrac{5}{3} = 350 \text{ lb.}$$

As 1 in. width of belting, $\frac{7}{32}$ in. thick, can safely withstand 70 lb. tension, the width of belting required $= \frac{350}{70} = 5$ in.

It will be seen from the example that the driving force is equal to the difference between the tensions on the tight and slack sides, and the same is true of every case of belt transmission of power. As 70 lb. is the safe working strength, and as with equal diameter pulleys one-half of the circumference, or 180 deg. of each pulley (both driver and driven), is embraced by the belt, the force that can be

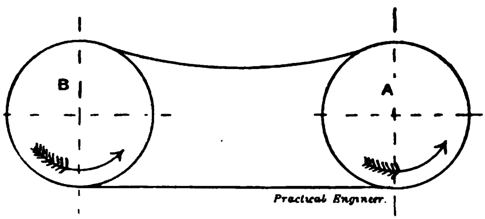


FIG. 41.

transmitted per 1 in. $= 70 \times \frac{3}{5} = 42$ lb. This value is sometimes taken at 50 lb., and if more or less than one-half of the circumference is embraced by the belt, the force transmitted may be increased or reduced by about $2\frac{3}{4}$ lb. for every 10 deg. difference from 180 deg.

As a substitute for leather, cotton and other forms of flat belting are frequently employed, and for fixed belts, not requiring to be shifted with the usual belt-shifting forks, they have met with considerable success. The principles of calculations for their application are precisely similar to those of leather belting above considered.

ROPE GEARING.

The transmission of power by fly ropes has become very popular in recent years, and is especially suitable for driving long line shafts on the various floors of factories engaged upon textile and other manufactures, where a very

large number of similar machines are employed, each requiring but a moderate amount of power to drive them. The usual method of driving is shown in fig. 42, representing an outline of an engine on the basement of the building, with a rope, or it may be several, running from the grooved

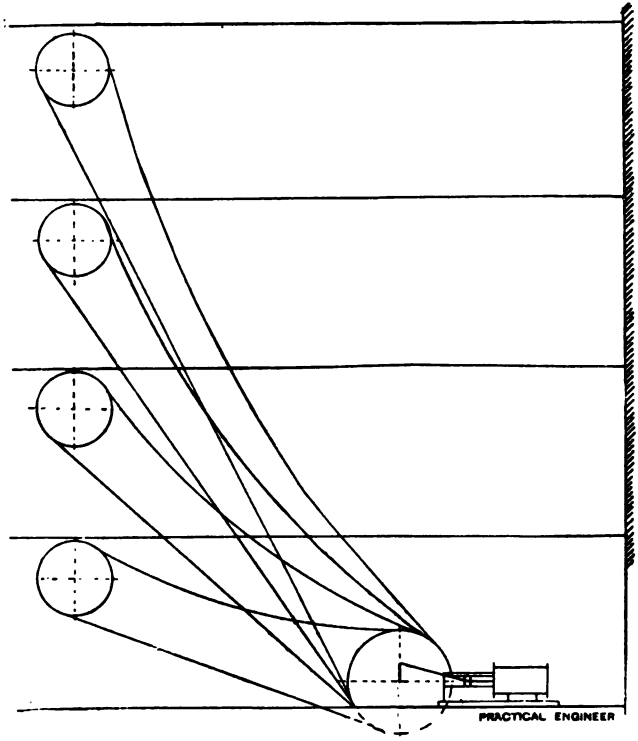


Fig. 42.

flywheel to grooved wheels secured to the line shaft serving each floor of the building. The usual form of the groove is shown at fig. 43. The ropes employed are sometimes of cotton, but hempen ropes are also very largely used of about $4\frac{3}{4}$ in. circumference, with $5\frac{1}{4}$ in. to $6\frac{1}{2}$ in. circumference for larger powers. In rope driving it is of great importance

that the slack or return side should be at the top, because the ropes must be run with the least possible initial tension. The diameter of the smallest pulley employed should not be less than 30 times the diameter of the rope, and considerably larger wherever possible. The life of the ropes will be much increased by the employment of large pulleys.

The fricton of a rope working in a taper groove as shown, on a cast-iron pulley, is three times greater than that of a rope working on a plain cast-iron pulley without a groove. The coefficient of the latter is ·28 ; hence with grooves free from grease the coefficient of friction between the ropes and the grooves will be $\cdot 28 \times 3 = \cdot 84$. With such a high coefficient the necessary initial tension on the slack side can be generally obtained solely from the weight of rope hanging between the driving and driven pulleys, and to this

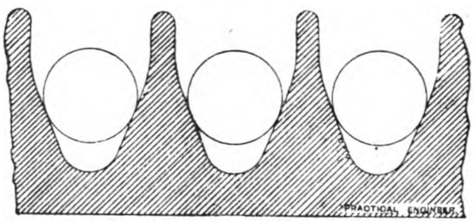


FIG. 43.

end the horizontal distance between the centres of the pulleys should be not less than 20 ft. The speed of the ropes varies from 3,000 ft. to 5,000 ft. per minute. Hutton gives the following simple rule for calculating the actual horse power that may be transmitted by rope gearing of the form described: Multiply 8 times the square of the circumference of one rope by the number of ropes, and by the circumferential velocity (or speed of rope) in feet per minute, and divide the product by 33,000.

Power may be also transmitted by wire ropes on what is known as Hirn's system. The ropes are of flexible wire $\frac{3}{8}$ in. to 1 in. diameter, and run at a speed of about 60 ft. per minute on grooved pulleys of from 12 ft. to 15 ft. diameter. The rope is not squeezed in the groove during running, as with hempen ropes, but rests on the bottom of the groove on hard wood let into the pulley. The system has so far been chiefly employed upon the continent and in America.

THE GAS ENG
F.C.S. Crown 8vo
post free 6s. 6d.

HOUSEHOLD B
and Prevention. I
of the Association for t
With numerous Illustra
Kitchen Boiler Apparat
Safety during Frost. Pr

TURNING LATH
and Apprentices.
post free 2s. 9d.

EXTI

"This is a work of 158 pa
matter on branches of lathe
these last it is valuable, as
work, how to avoid the m
certain kinds of tools; both
Engineer.

Demy

STEAM LOCOM
By WILLIAM F
Practically Considered."

EXTR

"To many only remotel
book must be of considerabl

"From beginning to end
it will become the standar
East Anglian Daily News.

SCREWS AND
account of the orig
and American Threads,
post free 2s. 9d.

EXTR

"The Britannia Compan
has led them to issue this lit
on screw-cutting to the man
thorough and practical com
valuable digest of all the
authorities at different times
to all those who are stud
construction."—*The Practica*

THE TECHNICAL PUBLISHING CO. LIMITED, 6, Victoria Station Approach, Manchester.
JOHN HEYWOOD, 2, Amen Corner, London; 33, Bridge Street, Bristol; and Booksellers.